

Theoretical and Experimental Investigation of Flow Straighteners in U-Type Pulse Tube Cryocoolers

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ABSTRACT

The U-type Pulse Tube Cryocooler (PTC) involves a change in the direction of gas flow as it proceeds from the regenerator to the pulse tube. The sharp U-bend has an adverse effect on pulse tube cooling action due to formation of eddies and undesirable mixing in the cold end of the pulse tube.

The present work deals with experimentation and CFD modeling related to U-type PTCs. Two cases of 'U' bends have been studied; gradual 'U' bends and sharp 'U' bends. Experimentation has been carried out using copper screens of 100-mesh size as flow straighteners. The optimum performance in terms of low temperature for the case of a gradual U bend was achieved with a stack of 18 flow-straightener screens. The no-load temperature for this case of a gradual 'U' bend, with and without flow straighteners, was 57.7 K and 88.8 K, respectively, for a charging pressure of 16 bar. When the gradual 180 degree bend at the cold end was replaced by a sharp U bend, the no load temperature increased from 88.8 K to 137 K without flow straighteners.

In order to understand the role of flow straighteners in PTC, a CFD model was developed in FLUENT. The flow straighteners are modeled as a homogeneous porous medium. The results show that the flow straighteners significantly affect velocity patterns in the pulse tube. The theoretical study showed that there exists an optimum number of flow straighteners which improve the cooling power and efficiency of the pulse tube cryocooler. This was in agreement with the experimental results obtained earlier.

INTRODUCTION

A pulse tube cryocooler is a vibration and maintenance free device that can attain and maintain a temperature below 80 K. There are various configurations possible for PTCs: like inline, U-type, coaxial, and annular. The gas enters the pulse tube from the regenerator, and turbulence in the gas takes place at the cold end of the pulse tube due to the sudden change in direction and gas flow area. In order to improve the performance of U-type PTC, it is essential to reduce the turbulence in flow of the gas entering the pulse tube. An effective way of reducing this turbulence is the use of flow straighteners. Kamoshita et al. [1] succeeded in improving the

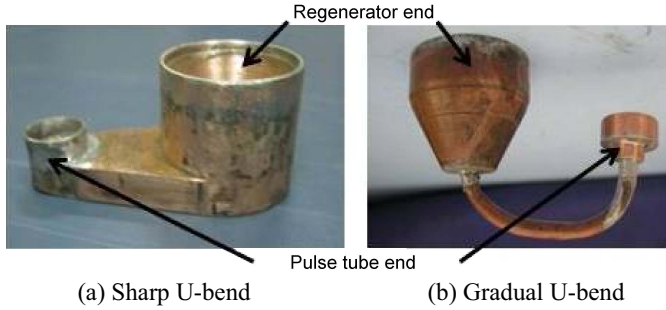


Figure 1 Photograph different U-type configurations

cooling power and efficiency of PTC just by adding flow straighteners at the cold end of the tube. Nachman et al. [2] analyzed the velocity and temperature field of oscillating flow around a 180 degree bend using CFD. Significant improvement is found in the uniformity of the flow field, resulting in a more uniform temperature field and improved performance of the PTC. Sudden enlargement of the gas flow at the entry of the pulse tube from the U bend results in formation of eddies and undesirable mixing of the gas inside the pulse tube. The flow straighteners reduce the turbulence and also the heat losses at the pulse tube cold end [3]. Liang and de Waele [4] analyzed the phenomenon of flow resistance of the flow straighteners and the viscous interaction with the tube wall resulting in flow streaming.

The present work strives to understand the effect of flow straighteners in two configurations of U-type PTCs as shown in Figure 1. The sharp U-bend configuration (Figure 1(a)) consists of two 90 degree bends, while the other configuration (Figure 1(b)) has a gradual U-bend. Experimental investigations were carried out for both configurations. Also, in order to understand the results, theoretical work using FLUENT was carried out to study the flow pattern at the cold end of the pulse tube.

CFD ANALYSIS

Figure 2 shows the configurations considered for the CFD analysis. The geometrical parameters are kept identical to the experimental work. The geometry consists of sharp or gradual U-bend (A), flow straighteners (B), pulse tube (C), hot end heat exchanger (D), Inertance tube (E) and reservoir (F).

Assumptions

The following assumptions were made to carry out the CFD analysis.

- A 2D model is considered to reduce computational time.
- The wall temperature of hot end heat exchanger, inertance tube, and reservoir is 300 K.
- Flow in the inertance tube is taken as turbulent.
- The temperature at the inlet of the U-bend from regenerator side is 55 K.
- The geometry is analyzed for compressible flow.

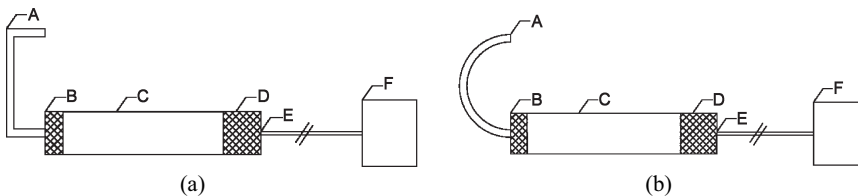


Figure 2. Schematic of the geometries (a) with sharp U-bend (b) with gradual U-bend.

Governing Equations

The overall governing equations employed for the analysis using FLUENT are given below [5].

Continuity equation

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{V}) = 0 \quad (1)$$

where ρ is the density of gas and \vec{V} is the velocity vector.

Momentum equation

$$\frac{\partial(\rho \vec{V})}{\partial t} + \nabla \cdot (\rho \vec{V} \vec{V}) = -\nabla P + \nabla \cdot (\bar{\tau}) \quad (2)$$

where P is pressure and $\bar{\tau}$ is the stress tensor and as given in equation (3)

$$\bar{\tau} = \mu \left[(\nabla \vec{V} + \nabla \vec{V}^T) - \frac{2}{3} \nabla \cdot \vec{V} I \right] \quad (3)$$

and I is the unit tensor

Energy equation

$$\frac{\partial(\rho E)}{\partial t} + \nabla \cdot [\vec{V} \cdot (\rho E + P)] = \nabla \cdot \left[k \nabla T + (\bar{\tau} \cdot \vec{V}) \right] \quad (4)$$

where E is energy.

The above governing equations are for a single fluid system, hence valid for all the components except the flow straighteners and the hot-end heat exchanger. The flow straighteners and hot-end heat exchanger are modeled as homogeneous porous media.

Governing Equations for porous media

The governing equations for porous media are as follows,

Continuity equation

$$\epsilon \frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{V}) = 0 \quad (5)$$

where ϵ is porosity.

Momentum equation

$$\frac{\partial(\rho \vec{V})}{\partial t} + \nabla \cdot (\rho \vec{V} \vec{V}) = -\nabla P + \nabla \cdot (\bar{\tau}) + S_I \quad (6)$$

where S_I is the source term, given by

$$S_I = - \left(\frac{\mu}{\alpha} V_i + C_2 \frac{1}{2} \rho V_{mag} V_i \right) \quad (7)$$

where μ is viscosity, α is permeability, and C_2 is an inertial resistance factor

The first term in the Equation 7 accounts for the viscous losses and the second term, for the inertial losses.

Energy equation

$$\frac{\partial(\epsilon \rho E_f + (1-\epsilon) \rho_s E_s)}{\partial t} + \nabla \cdot [\vec{V} \cdot (\rho E_f + P)] = \nabla \cdot \left[K_{eff} \nabla T + (\bar{\tau} \cdot \vec{V}) \right] \quad (8)$$

Table 1 Operational parameter of the porous media

Material	m	$n \text{ (m}^{-1}\text{)}$	$dw \text{ (m)}$	$l \text{ (m)}$	β	ϵ	D	C_2
Copper	100	8.74×10^3	1.14×10^{-4}	1.4×10^{-4}	0.367	0.697	2.42×10^9	1.56×10^4

where K_{eff} is the effective thermal conductivity, computed by volume average of fluid and solid conductivity

$$K_{eff} = \epsilon k_f + (1 - \epsilon) k_s \quad (9)$$

In FLUENT, the system is modeled using a two-dimensional segregated unsteady solver. The second-order upwind convection scheme is chosen to solve the momentum and turbulence kinetic energy equations for discretization. The standard k- ϵ turbulence model is used for turbulent flow. The PISO algorithm is selected for pressure-velocity decoupling. All boundary conditions except at the entry of the sharp U-bend are standard as defined in FLUENT. A pressure oscillating boundary condition is given at the entry of the sharp U-bend using a user-defined function (UDF). The pressure at the U-bend inlet is given by

$$P = P_0 + \sin(\omega t) \quad (10)$$

where ω is angular velocity.

The following one-dimensional empirical equations are considered for the pressure gradient in the porous media [4].

$$\frac{dp}{dx} = -f_F \frac{2\rho_f V^2}{d_h} \quad (11)$$

$$f_F = \frac{n \epsilon d_h^2}{4l\beta} \frac{33.6}{\text{Re}_h} + 0.337 \frac{n \epsilon^2 d_h}{4\beta^2} \quad \text{Re}_h = \frac{\rho V d_h}{\mu} \quad (12)$$

where l is the mesh distance, β is the opening area ratio of screen, d_h is the hydraulic diameter of screen, n is the number of packed screens per unit length, μ is the viscosity. β can be calculated as

$$\beta = \frac{(x_t - 1)^2}{x_t^2} \quad (13)$$

where

$$x_t = \frac{0.0254}{d_w m} \quad (14)$$

and m is mesh per inch, d_w is wire diameter of screen. The operational parameters of the porous media are given in Table 1.

EXPERIMENTAL SETUP

Experiments were carried out on a split type PTC. The operating frequency was kept constant at 50 Hz, while the system was kept at a charging pressure of 16 bar (gauge) using helium as the working fluid. The regenerator and the pulse tube lengths were 54 mm and 74 mm, respectively. The diameter of regenerator and pulse tube were kept as 28 mm and 12.2 mm, respectively, with a wall thickness of 0.15 mm. SS screens of 400-mesh number were used as the regenerator matrix material. The cold end is made of two different geometries. The inner diameter of the gas passage in the copper block with the sharp U-bend was kept at 3.5 mm. In the case of the gradual U bend, the inner diameter of the copper tube was kept the same as the one with the sharp bend. However, the length of the tube in this case was around 10 mm longer.

Coarse copper screens of 100-mesh size were used as the flow straighteners to minimize the losses associated with pressure drop.

ENDEVCO Piezo-resistive transducers were used for pressure measurement, while a Yokogawa DL 750P oscilloscope was used for dynamic pressure wave display and data collection. Temperatures were measured with the help of silicon diodes.

RESULTS AND DISCUSSIONS

A. Theoretical Results

The numerical analysis was carried out using FLUENT to understand the effect of flow straighteners on the velocity and temperature profile at the inlet of the pulse tube.

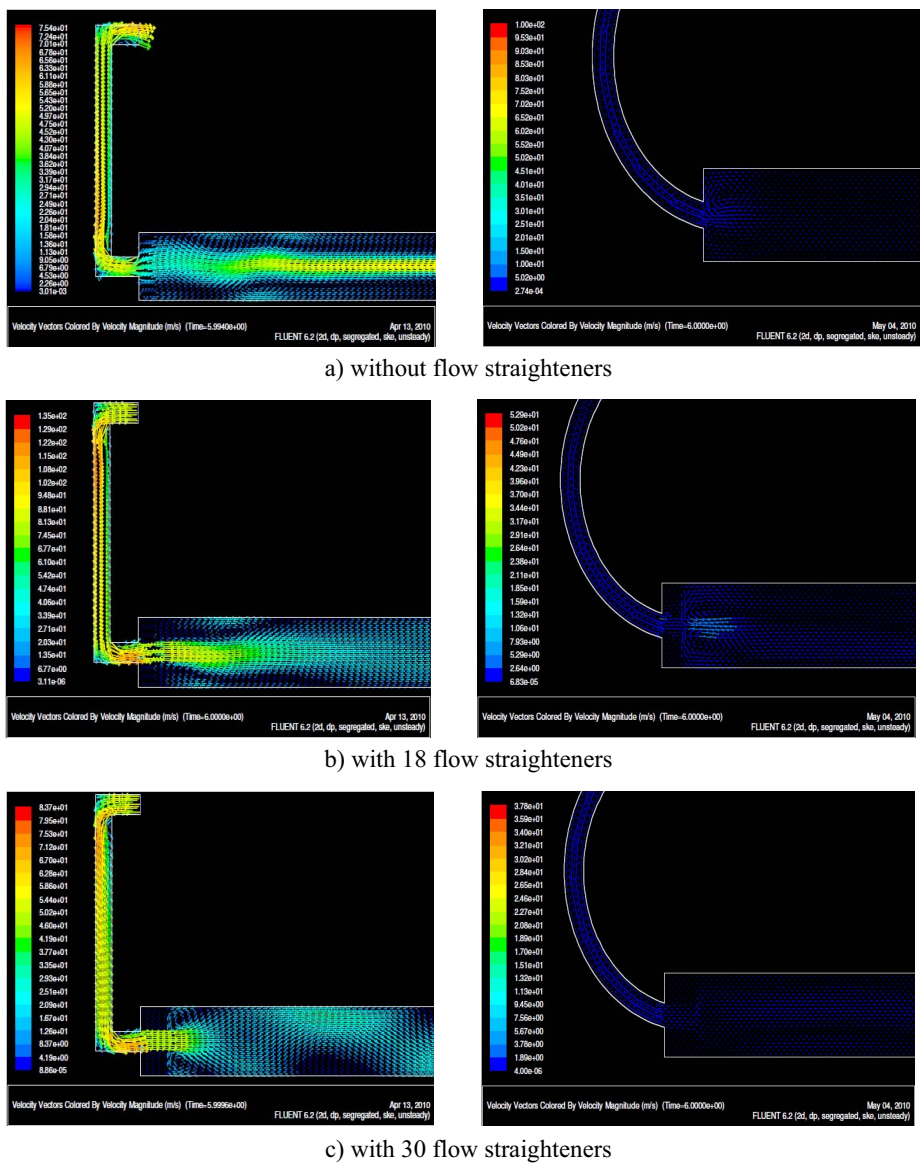


Figure 3 Velocity contours for sharp and gradual U bend configurations

Figure 3 shows the velocity contours inside the pulse tube for different numbers of flow straightener screens. The whirling flow pattern at the cold end of the pulse tube is visible in the case of zero flow straighteners (Figure 3(a)) as well as with 30 flow-straightener screens (Figure 3(c)). However, with 18 flow-straightener screens the whirling action moves away from the cold end of the pulse tube (Figure 3(b)). This may be attributed to the inertia and viscous effect due to the flow straighteners. The temperature gradient in the gas along the pulse tube length is observed. The minimum temperature observed in the analysis for zero screens, 18 screens, and 30 screens was 274 K, 120 K and 160 K, respectively, for the sharp U-bend configuration. The temperatures in the case of the gradual U bend were 95 K, 55 K and 55 K, respectively. The results obtained are for truncated CFD simulations due to time limitations. However, they signify the importance of flow straighteners especially in the sharp U-bend configuration.

B. Experimental Results

Figure 4 shows the effect of number of screens on the performance of the PTC, as defined in terms of temperature at the cold end of pulse tube. It can be observed that the PTC with the sharp U bend reached a temperature of 137 K with zero flow straighteners. This is due to a significant amount of turbulence in the gas flow. The performance improved with increased numbers of flow-straightener screens at the cold end of the pulse tube, up to 18 screens. The minimum temperature reached was 64 K with 18 screens. Increasing the number of screens from 18 to 30 degraded the performance, which is reflected by the temperature rising about 7 K as compared to the earlier case.

The copper block with the sharp turn was next replaced by the one with the smooth bend. This configuration of PTC reached a lowest temperature of 88.5 K with zero flow straighteners. The minimum temperature reached was 57.8 K with 18 screens. Increasing the number of screens from 18 to 25 degraded the performance, and the temperature rose 1.3 K. This is due to the increased pressure drop losses. Compared to the sharp U bend, better performance is observed in the case of the gradual U bend due to the reduction in losses associated with the sudden change in gas flow direction. Figure 3 offers a comparison of the velocity contours, which justifies the low temperature obtained using the gradual U bend.

Figure 5 shows a comparison between the theoretical and experimental results for the cold end temperature in the case of the sharp U bend (Figure 5a) and for the gradual U-bend (Figure 5b). The numerical predications and the experimental findings show similar trends. The results based on CFD analysis, as specified earlier, are based on truncated computer simulations.

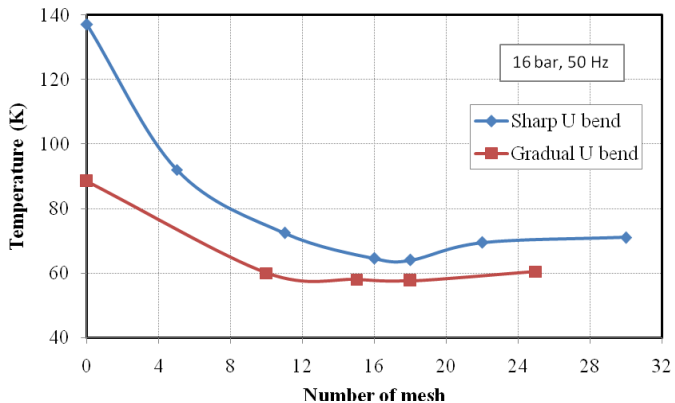


Figure 4 Effect of flow straighter on minimum temperature

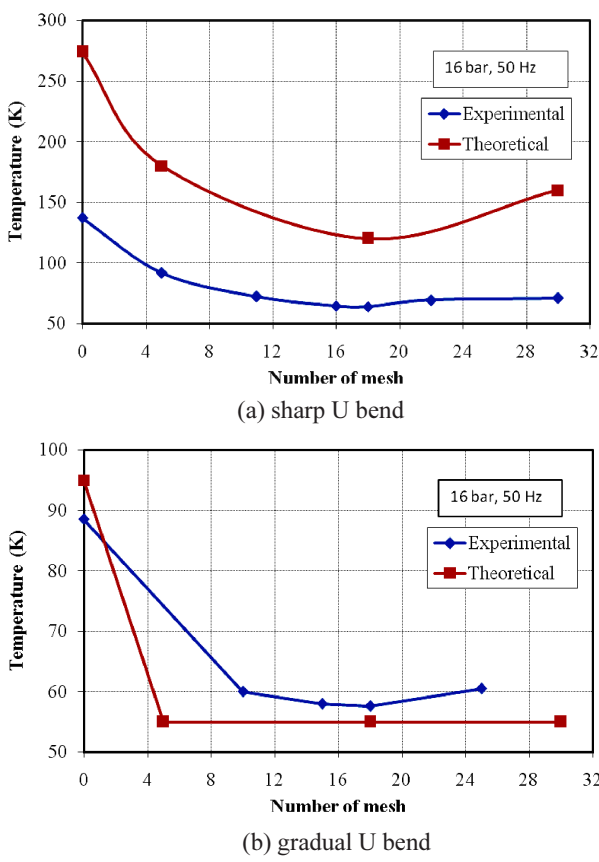


Figure 5. Comparison between experimental results and CFD analysis.

CONCLUSION

The U-bend, together with the sudden change in cross section area at the cold end of the pulse tube, have an adverse effect on pulse tube cooling due to formation of eddies and undesirable mixing at the cold end of the pulse tube. The theoretical analysis and experimental investigation have demonstrated that flow straighteners are very effective at reducing the formation of eddies in U-type PTCs. There exist an optimum number of flow straighteners; the deviation from which on either side results in degraded performance.

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